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Comparison study of solar cooling technologies for an institutional building in subtropical Queensland, Australia

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ABSTRACT

In the last few years, due to soaring fuel prices and gas emissions, buildings designers have suggested new cooling technologies that can compete with conventional cooling technologies. The interest in solar assisted cooling technologies stems from the fact that they are sustainable and environmentally friendly compared to conventional cooling systems. The present paper investigates and compares the performance of two types of solar cooling: solar absorption cooling system and solar desiccant cooling system, for an institutional building located in the state of Queensland, Australia. A simulation model of the building has been developed for the evaluation of cooling load and primary energy usage with the application of the solar cooling systems. The evaluation of these systems has been conducted taking into account the solar fraction, the coefficient of performance and the primary energy savings. The building cooling load profile and the technologies component were modelled using TRNSYS16 in Subtropical Central Queensland typical meteorological year. Results showed that by installing a 50 m^2 of solar collectors, the desiccant system will achieve a maximum of 0.89 and 0.82 of solar fraction in the month of December and January in Gladstone and Rockhampton sites respectively, while the absorption system will achieve a 0.81 and 0.75 of solar fraction for the same time period in Gladstone and Rockhampton sites respectively.

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Contents

1. Introduction

Due to extensive usage of air conditioning in the state of Queensland (Australia) on hot summer days, the electricity grid increasingly faces the danger of overloading which might lead to service disruption, economic and environmental threats. In the

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light of this, the possibility of using a renewable energy to drive a cooling process such as solar energy has been raised due to the abundance of sunshine in Queensland subtropical areas. Solar assisted air conditioning is a promising alternative to conventional mechanical HVAC systems as this technology proved to be a success in a number of demonstration projects around the world and has the ability to reduce greenhouse gas emissions.

A number of studies have assessed solar cooling technology based on its economic analysis and energy performance. Fong et al. have compared different cooling technologies under Hong Kong hot and humid climate [1]. Pongtornkulpanic et al. have investigated the feasibility of solar absorption cooling under Thailand climate [2]. Eiker and Pietruschka have presented a full simulation model for absorption cooling systems under European climate [3]. Tsoutsos et al. have presented a comparative study for solar cooling economic evaluation of two types of solar cooling: absorption and an adsorption system under Greek climate [4]. Koroneos et al. have investigated the applicability of different solar cooling technology for a medical centre [5]. Hartmann et al. have presented a comparison of solar thermal and photovoltaic cooling technology for two different European climates [6]. Relatively there are a few studies that have been undertaken in Australia to investigate the technical and economical comparison between solar thermal cooling technology. Leutz et al. have investigated sorption cycles for air conditioning applications for climates prevalent in Australasia [7]. White et al. have investigated the performance of solar desiccant cooling system in warm temperature climates of Melbourne, Sydney and Darwin [8]. Alizadeh has investigated experimental behaviour of solar cooling under summer conditions of Adelaide and Brisbane in two different studies [9,10]. It is very important for researchers to focus on using different thermally driven cooling technologies and to provide a precise comparison between the most two commercially available solar cooling technologies: solar absorption cooling and solar desiccant cooling. An extensive evaluation and comparison of solar cooling technologies; solar desiccant cooling and solar absorption cooling to assess the potential of solar assisted air conditioning in Central Queensland's region, Australia in terms of solar fraction, coefficient of performance, primary energy usage and energy savings presented in this study. Furthermore this paper presents a mathematical assessment of two solar cooling technologies under two different Subtropical Queensland climates; namely Rockhampton and Gladstone with reference to a conventional cooling system.

2. Solar assisted air conditioning

Generally cooling system rely on electric energy to drive its cooling process which causes a significant negative impact on the environment as 90% of Australia’s electricity is generated from fossil fuel [11]. In the region of Central Queensland the average annual temperature has increased 0.5 °C in the last 10 years and will reach up to 4.5 °C by the year 2070 [12]. These reasons will force policy makers and developers to adopt a non conventional cooling technologies that have less negative impact on the environment and economically viable. Vast numbers of researchers and experts consider solar assisted air conditioning as the future alternative of conventional cooling methods since this new technology proved to be sustainable, feasible, environmental friendly, and economically viable. The existing operated solar assisted air conditioning systems around the world can be classified into solar electric cooling technology and solar thermal cooling technologies. Solar electric cooling technology uses a conventional electric vapour compressor air conditioning process as an electrical energy is provided by solar photovoltaic (PV) panels. In solar thermal cooling technologies, solar heat is required to drive the cooling process. This can be done by collecting solar radiation using thermal solar collectors to convert it into thermal energy, and use this energy to drive thermally driven cooling cycles such as desiccant, absorption and adsorption cycles. This paper focuses on solar thermally driven cooling technologies: absorption and desiccant as these technologies have a better economical and technical performance than solar photovoltaic cooling technologies.

2.1. Solar absorption cooling technology

According to [13,14] absorption chillers are the most used thermally driven cooling system and the most dominant technology in solar cooling application. The thermodynamic cycle of absorption chillers' is normally driven by a heat source. In short the main concept behind the function of absorption cycle is that a fluid pump is utilised to circulate at least two chemical components; the refrigerant, and the sorbent. In some sorts of commercially available absorption chillers required to produce water temperature between 5 °C and 8 °C, the refrigerant is water and the absorbent is lithium bromide (LiBr). While using ammonia (NH_3) as a refrigerant and water as an absorbent suites special industrial refrigeration needs and those applications that require water temperature under 5 °C [15,16]. The function of the two types of chillers is similar to conventional cooling system where the role of the mechanical compressor is replaced by what is known as a thermal compressor which consists of an absorber, a generator, a pump, a condenser, an evaporator and a circulating valve as shown in Fig. 1. The absorption cooling cycle starts in the evaporator where the refrigerant evaporates in a low partial pressure environment to the absorber, the evaporation of the refrigerant will causes heat to be extracted from surroundings and cool down the chilled water. Then the gaseous refrigerant is absorbed into other liquid (the absorbent) which causes its partial pressure to be reduced in the evaporator and allowing more liquid to evaporate. The diluted liquid of the refrigerant and the absorbent materials is pumped to the generator where the mixture liquid is heated causing the refrigerant to evaporate and then condensed on the condenser (heat exchanger) to refill the supply of liquid refrigerant in the evaporator through a circulation valve. Thus the energy used to power the pump is neglected when compared to the energy required by a conventional compressor [17,18].

2.2. Solar desiccant cooling

There are limited techniques that can be used to reduce electricity consumption by conventional air-conditioning. Among those techniques, solar desiccant cooling is considered as an

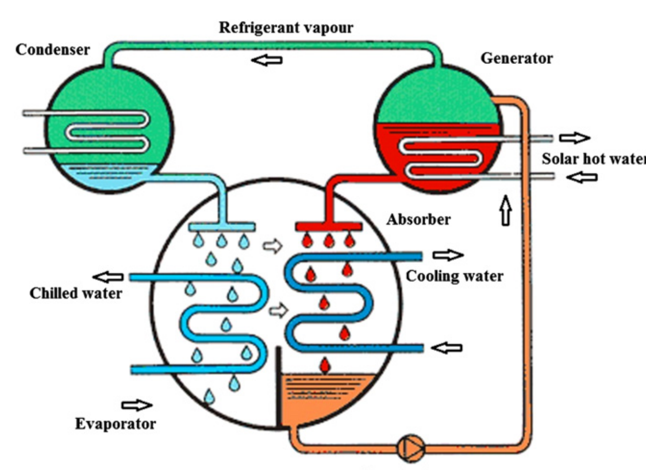

Fig. 1. A schematic of absorption chiller [19].

attractive alternative to current conventional cooling systems. In the past decade many researches focused on applying this cooling technology since it uses low grade thermal energy resources which can be delivered by solar collectors, and also can help to eliminate the usage of refrigerant gases like Chlorofluocarbons (CFCs) [20]. Solar desiccant cooling is more effective when used in hot and humid climates since the prime feature of desiccant cooling system that they can treat sensible and latent heat load separately. In general solar desiccant cooling technology consists of three sub-systems: solar thermal system, desiccant dehumidifier and evaporative cooler. The desiccant dehumidifier can be controlled separately using a humidistat that measures and controls the humidity and latent load of a cooled space. The evaporative cooler can be controlled using a thermostat to measure and control the sensible load. However in the desiccant cooling process main concept depends on the desiccant materials ability to reduce the air moisture content for air cooling process and dehumidification. As a result of this constant air drying; the dehumidified air is heated above ambient temperature. In order to reduce air sensible temperature, there are cheap air cooling techniques, like evaporative cooling or heat exchanger which are used to cool the heated dry air to near ambient temperature as shown in Fig. 2.

3. Technical performance indicators

In solar assisted air conditioning there are a numbers of technical performance indicators. These indicators are solar fraction, coefficient of performance, solar energy gain (energy savings), and primary (parasitic) energy used. In this study a year round simulation results in prospective to the following indicators is used to assess the potential of each type of solar cooling system in each climate.

3.1. Solar fraction

Solar fraction is considered as the most used technical indicator in order to evaluate the performance of solar cooling systems. Solar fraction (*SF*) measures the ratio of thermal energy produced by the solar collectors to the cooling system total driving energy [21]. Solar fraction depends on many factors such as load, collector's area, hot water storage size, and solar radiations availability. When the solar thermal energy is insufficient to drive the cooling process, a backup heater is used to deliver the required energy. Therefore solar fraction (*SF*) can be expressed as in the following equation [22]:

$$SF = \frac{Q_{Solar}}{Q_{Solar} + Q_{aux}} \tag{1}$$

where Q_{Solar} is the thermal energy produced by solar collectors in kW and Q_{aux} is the thermal energy produced by the auxiliary heater in kW.

3.2. Coefficient of performance (COP)

Coefficient of performance (*COP*) is a general cooling system (HVAC) performance indicator. *COP* is defined as the ratio of cooling amount produced by cooling system (HVAC) to the total energy consumed by the cooling system. The cooling system with high *COP* is more efficient than the ones with lower *COP*. The coefficient of performance (*COP*) for a conventional system (Vapour compression) is defined by the following equation:

$$COP = \frac{Q_{Ce}}{W_{el}} \tag{2}$$

where Q_{Ce} is the refrigeration effect in kW and W_{el} is the cooling system total electric power input in kW.

For solar absorption cooling system, the coefficient of performance COP_{Abs} is as in the following equation:

$$COP_{Abs} = \frac{Q_{chw}}{Q_{Aux} + Q_{Solar} + W_{el\text{-}primary}} \tag{3}$$

where, Q_{chw} is the energy removed from chilled water stream in kW, Q_{Aux} is the energy drawn from the auxiliary heater in kW, Q_{Solar} is the thermal energy produced by the solar collectors in kW, and $W_{el\text{-}primary}$ is the total electric energy used by the cooling system like pumps in kW. To calculate solar thermal energy produced by the solar collectors the following equation is used.

$$Q_{solar} = \eta_{solar} \times A_a \times G \tag{4}$$

where Q_{solar} is the Solar radiation in the collector plane in kW, A_a is the Aperture area of the collector in m^2 and G is the Solar radiation in the collector plane in kW/m^2.

Solar collector's efficiency is defined as in the following equation:

$$\eta_{solar} = \eta_0 - C_1 \times \frac{t_m - t_a}{G} - C_2 \times \frac{(t_m - t_a)^2}{G} \tag{5}$$

where η is the collector efficiency, η_0 is the optical efficiency, C_1, C_2 are the collector heat loss coefficient, t_m is the collector temperature and t_a is the ambient temperature.

Then solar thermal energy produced by the collectors is given bythe following equation:

$$Q_{solar} = \left(\eta_0 - C_1 \times \frac{t_m - t_a}{G} - C_2 \times \frac{(t_m - t_a)^2}{G} \right) \times A_a \times G \tag{6}$$

The auxiliary energy needed to drive the cooling process when solar thermal energy is not sufficient can be calculated as

$$Q_{Aux} = Q_{loss} + Q_{fluid} = \frac{m.C_w(T_{set} - T_{in}) + U_A(\overline{T} - T_{env})}{\eta_g} \tag{7}$$

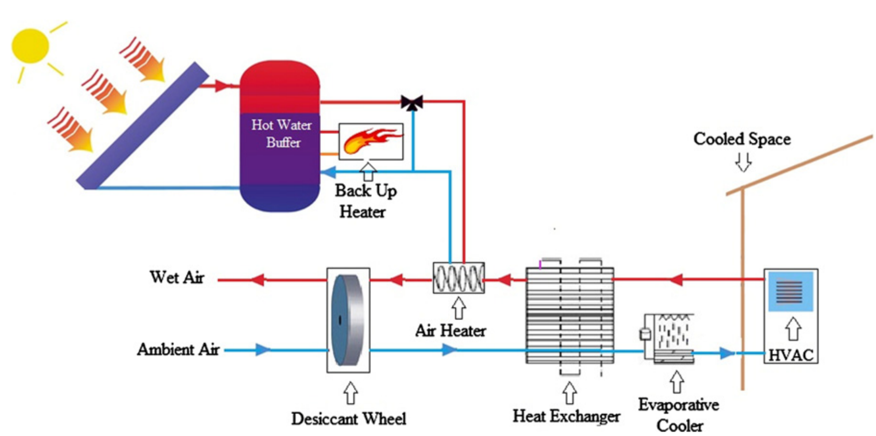

Fig. 2. A schematic of solar desiccant cooling system.

where Q_{Aux} is the required heating rate including efficiency effects of the backup heater in kW, η_g is the backup heater efficiency, Q_{fluid} is the rate of heat addition to water stream, m is the inlet water mass flow rate, T_{set} is the set temperature of heater internal thermostat in °C, T_{in} is the water inlet temperature in °C, U_A is the overall loss coefficient between the backup heater and its surroundings during operation, $\overline{T}$ is the $(T_{set}+T_{in})/2$ and T_{env} is the temperature of heater surroundings for loss calculations in °C. Then solar absorption cooling system coefficient of performance COP_{Abs} is calculated by the following equation:

$$COP_{Abs} = \frac{Q_C}{((\eta_0 - C_1(t_m - t_a/G) - C_2((t_m - t_a)^2/G))A_aG) + ((m.C_w(T_{set}\_T_{in}) + U_A(\overline{T} - T_{env}))/\eta_g) + W_{el\text{-}primary}} \tag{8}$$

For solar desiccant cooling system, the coefficient of performance COP_{desi} can be given by the following equations:

$$COP_{desi} = \frac{Q_C}{Q_{regen} + Q_{evap}} = \frac{m_a(h_o - h_s)}{Q_{solar} + Q_{Aux} + Q_{evap}} \tag{9}$$

$$COP_{desi} = \frac{m_a(h_o - h_s)}{((\eta_0 - C_1(t_m - t_a/G) - C_2((t_m - t_a)^2/G))A_aG) + ((m.C_w(T_{set}\_T_{in}) + U_A(\overline{T} - T_{env}))/\eta_g) + W_{el\text{-}primary} + Q_{evap}} \tag{10}$$

where Q_{evap} is the energy consumed by the evaporative cooler in kW, m_a is the mass flow of the regeneration air in kg s^{-1}, and $(h_o - h_s)$ is the enthalpy difference between outside air and supply air.

3.3. Primary energy consumption and energy saving

This technical indicator considers the overall primary and parasitic energy used by each system. Primary energy is the total electric energy used by the main system components or by the system auxiliary components.

In case of solar absorption cooling system the total electric energy used by the system can be expressed by the following equation:

$$W_{el\text{-}primary} = \left(\frac{m.C_w(T_{set}\_T_{in}) + U_A(\overline{T} - T_{env})}{\eta_g}\right) + Q_{Parasitic} \tag{11}$$

where η_g is the backup heater efficiency and $Q_{parasitic}$ is the total parasitic energy used by the system main and auxiliary components in kW.

In case of desiccant cooling system it is a similar scenario to absorption cooling system however the different is based in the usage of an evaporative cooler as given in the following equation:

$$W_{el\text{-}primary} = \left(\left(\frac{m.C_w(T_{set} - T_{in}) + U_A(\overline{T} - T_{env})}{\eta_g}\right) + \left(\frac{Q_{evap}}{\eta_e}\right)\right) + Q_{Parasitic} \tag{12}$$

where η_e is the evaporative cooler efficiency. The potential of energy saving E_{saved} for producing a certain cooling power can be evaluated based on the comparison between a conventional cooling system and a solar cooling system as in the following equation in kW:

$$E_{Saved} = \frac{(W_{Conv}/Q_{C,Conv}) - (W_d/Q_{C,D})}{(E_{Conv}/Q_{C,Conv})} \times L_{total} \tag{13}$$

where E_{Conv} is the conventional system electric power, $Q_{C,Conv}$ is the conventional cooling system capacity in kW, E_{Solar} is the solar cooling system electric power in kW, $Q_{C,Solar}$ is the solar cooling system cooling capacity in kW and L_{total} is the conditioned building total cooling load in kW.

4. Central Queensland subtropical climate

Australian Central Queensland main regions are Rockhampton region, represented by Rockhampton city and Gladstone region represented by Gladstone city. The city of Rockhampton located at −23.4°N (Latitude) and 150.5°E (longitude) and the city of Gladstone is located at −23.9°N (Latitude) and 151.3°E (longitude). The region Subtropical climate characterised by long, hot and humid summer with a high average due point. Fig. 3 shows the mean, minimum and maximum temperature of Rockhampton city. Throughout the summer Rockhampton maximum temperature ranges between 29 °C and 36 °C. Gladstone summer maximum temperature ranges between 26 °C and 34 °C as in Fig. 4 [11,23].

5. System design and assumptions

In this paper, a proposed building similar to building 41, the health and safety office at Central Queensland University Rockhampton campus was evaluated. Similar building and its parameters are proposed as well for Gladstone subtropical climate. Software TRNSYS 16 is used to calculate the building cooling load in both locations. The building is 3 m height and its total area is 128 m^2 as shown in Fig. 5. The building is modelled as a single zone building with parameters as shown in Table 1.

In this paper, a comparison is done between two solar cooling techniques: absorption cooling and desiccant cooling with reference to a conventional cooling system. Solar cooling systems consist of two main sub systems; solar sub system and the cooling sub system. Table 2 shows the model components which is called Types used in TRNSYS simulation studio. The model used a flat plate collectors Type 1b, a thermal storage tank Type 4c, a backup

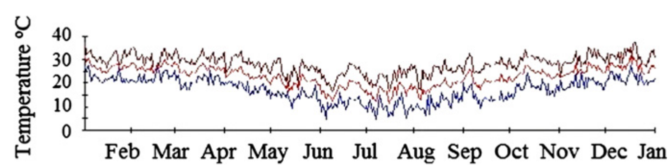

Fig. 3. Rockhampton daily temperature values (mean, minimum and maximum).

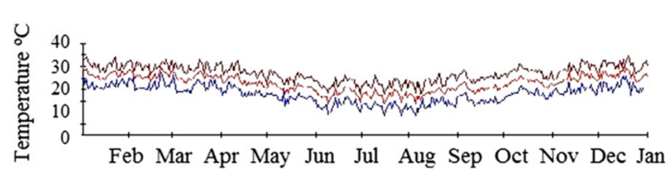

Fig. 4. Gladstone daily temperature values (mean, minimum and maximum).

Fig. 5. Building model.

Table 1
Building parameters.

Fraction of windows in external walls, north, west, east and south	10%, 25%, 25% and 50%	Door height and width	0.9 m and 1.8 m
Occupants density is	$0.1/m^2$	External brick thickness	110 mm
Computers	7×230 W	Air Cavity thickness	50 mm
Illumination	$2\ W/m^2$	Insulation	Sisalation foil
Specific gain	$14\ W/m^2$	Timber frame thickness	90 mm
Fresh air flow rate	10 L/Person	Plasterboard thickness	10 mm
Infiltration rate	0.2 Vol/h	Footing	Concreter
Windows height and width	1.8 m and 5 m	Roof	Conventional

Table 2
TRNSYS types.

Type 109	Reading weather data in TMY format	Type 6	Absorption chiller
Type 56a	Models the thermal behaviour of a building	Type 107	Single speed pump
Type 33e	Dry bulb temperature and relative humidity of moist	Type 654	Cooling loads imposed on a flow stream
Type 2d	ON/OFF differential controller	Type 682	Cooling tower
Type 69b	Effective sky temperature radiation exchange	Type 51b	Overhang and wing wall shading
Type 65d	Online graphical plotter	Type 34	ON/OFF humidity controller
Type 25c	Output file printer	Type 658	Room thermostat
Type 3b	Pump	Type 645	Annual forcing function
Type 1b	Flat-plate solar collector	Type 519b	Evaporative cooling device
Type 4c	Storage tank	Type 506d	Heat exchange
Type 11 h	pipe tee-pieces	Type 650	Rotary desiccant wheel
Type 6	Auxiliary heater	Type 683	Absorption chiller

electric heater Type 6, an absorption chiller Type 107, a desiccant wheel Type 683 and other piping, pumping and ducting components identified in Table 2. Tables 3, 4 and 5 show Types, certain specifications and parameters.

6. Building cooling load

Cooling load is known as the total amount of heat energy removed from a space using cooling mechanism in a unit time. Cooling load is equal to the rate at which heat is generated by solar radiation, infiltration, ventilation, people, machinery, and processes, as well as the net flow of heat into the space not associated with the cooling machinery. Building cooling load is normally assessed and calculated based on summer cooling demand.

Heat gain is divided into sensible and latent: Sensible heat is the heat content causing an increase in dry bulb temperature whereas latent heat is the heat content due to the presence of water vapour in the atmosphere. Sensible and latent heat gain has internal gain source and external gain source originated by solar radiation, heat transfer through opaque building envelop, internal heat gain, ventilation, infiltration and heat transfer between the building and surroundings.

Building parameters are governed by two thermal balance equations. Eq. (14) due to sensible heat gain and Eq. (15) due to latent heat gain [24].

$$\frac{dT}{dt} = \frac{UA}{cap}(T_{amb}-T_{set}) + \frac{m_{vent}C_{pair}}{cap}(T_{vent}-T_{set}) + \frac{m_{\inf}CP_{air}}{cap}(T_{\inf}-T_{set}) + \Sigma Q_{gain} \quad (14)$$

Total building sensible heat gain is calculated using the following equation:

$$\Sigma Q_{gain} = Q_{light} + Q_{equip} + Q_{people} + Q_{\inf} + Q_{vent} + Q_{solar} \quad (15)$$

where T is the zone temperate (°C), U is the building loss coefficient in kW/h m^2 C, A is the space over all area in m^2, cap is the building capacitance in kW/C, T_{set} is the Space set temperature in °C, $\dot{m}_{vent}$ is the ventilation air mass flow rate in kg/h, C_{pair} is the specific heat of building air in kW/kg °C, T_{vent} is the temperature of ventilation air in °C, $\dot{m}_{\inf}$ is the mass flow rate of infiltration air in kg/h, T_{inf} is the temperature of infiltration air in °C, Q_{gain} is the

Table 3
Flat plate collectors parameters (Type 1b) and hot water storage tank parameters (Type 4c).

Conversion factor $\eta 0$	0.780	volume of the stored fluid per unit of collector area	70 l/m^2
Lost coefficient C_1	4.2 w/m^{2}K	Collector fluid flow rate per unit area	0.015 kg/sm^2
Lost coefficient C_2	0.008 w/m^{2}K^2	Hot water storage tank loss coefficient	0.5 W/m^2 K

Table 4
Single affect absorption chiller parameters Type 107.

COP	0.7	Hot water flow rate	0.12 m^3/h/KW
Cooling power	10 KW	Cooling water temperature (Inlet/Outlet)	27/35 °C
Chilled water temperature (Inlet/Outlet)	6/15 °C	Cooling water flow rate	0.26 m^3/h/KW
Chilled water flow rate	0.29 m^3/h/KW	Electrical power	20 KW
Hot water temperature (Inlet/Outlet)	72/62 °C	Hot water flow rate	0.12 m^3/h/KW

Table 5
Desiccant wheel parameters (Type 683) and evaporative cooler Type (506d) parameters.

Capacity	10 Kg/h	Process air evaporative cooler saturation efficiency	90%
Nominal dry air flow	1450 m^3/h	Exhaust air mass flow rate	3.36 Kg s^{-1}
External static pressure	100 Pa	Evaporative cooler saturation efficiency	75%
Nominal wet air flow	580 m^3/h	Evaporative cooler power consumption	0.1 KW
Speed of rotor	42 r/h	Desiccant wheel power consumption	0.2 KW

sensible heat gain in kW, Q_{lihgt} is the heat gain resulted from lights, Q_{equip} is the heat gain resulted from equipment used in the building including computers and printers in kW, Q_{people} is the sensible heat gain resulted from people in kW, Q_{inf} is the sensible heat gain from infiltration in kW, Q_{vents} is the sensible heat gain due to ventilation and Q_{solar} is the heat gain due to sun radiation in kW.

Eq. (16) governed building heat balance due to latent heat gain.

$$\frac{d\omega}{dt} = \frac{\dot{m}_{\inf}}{\rho V}(\omega_{vent} - \omega) + \frac{\Sigma\omega_{gain}}{\rho V} \tag{16}$$

Total building sensible heat gain is calculated using the following equation:

$$\Sigma\omega_{gain} = \omega_{people} + \omega_{\text{infl}} + \omega_{vents} + \omega_{equip} \tag{17}$$

where ω is the heat gain due to air moisture content in kW, ρ is the density of building air in kg/m^3, V is the building volume in m^3, ω_{gain} is the total heat gain due to moisture content on the air (latent), ω_{people} is the latent heat gain from people, ω_{inf} is the latent heat gain from infiltration, ω_{vent} is the latent heat gain due to ventilation and ω_{equip} is the latent heat gain using equipment and cooking.

7. Results and discussion

The combination of Central Queensland sunny climate and latitude gives it a substantial potential for solar energy production. Most of the Central Queensland region receives in excess of 5.55 kWh/m^2/day of solar radiation energy the whole year around. As shown in Fig. 6. Gladstone city recorded 6.5 kWh/m^2/day in the month of October, followed by the month of December and then January recording 6.1 kWh/m^2/day and 5.6 kWh/m^2/day respectively. Rockhampton city peaked in the month of November recording an average of 5.4 kWh/m^2/day followed by the month of December and then the month of January recording 5.1 kWh/m^2/day and 4.8 kWh/m^2/day respectively.

Central Queensland region high cooling demand is January to April and September to December as demonstrated in Fig. 7. Rockhampton proposed building recorded 6428 kWh of annual cooling load while Gladstone proposed building cooling load was 6750 kWh. The maximum cooling demand for Rockhampton proposed building was in the month of December at 810 kWh followed by the months of November, March, February, January, October, April, September, May, August, June and July. It was a similar story for the proposed building in Gladstone which recorded a maximum cooling demand in the month of December at 850 kWh followed by the months of November, March, February, January, October, April, September, May, August, June and July.

In the case of solar cooling system, solar collectors convert sun radiation into thermal energy. The resulted thermal energy is used to drive the cooling process. A backup electric or gas heater which normally fitted within the solar system is used when solar gained energy is insufficient to drive the cooling process. However, using absorption cooling technology in Rockhampton and Gladstone site covers the cooling load demand completely despite the fact of sensible and latent load differences. A desiccant cooling technology will deal with the latent cooling load only. Fig. 8 shows the solar gained energy versus the backup heater energy in the two proposed building using absorption solar cooling technology and desiccant cooling technology. It is noted that using absorption cooling system will require more driving energy (Solar energy and Backup heater energy) as the system will deal with 100% of the cooling load (Latent and Sensible) while desiccant cooling system requires less driving energy (Solar energy and Backup heater energy) as the system deals with nearly 50% of the cooling load (Latent load only). Furthermore it was observed that the gained solar energy is directly proportional to the solar collector's area.

Figs. 9 and 10 show the average monthly coefficient of performance (*COP*) of solar absorption cooling technology and solar desiccant cooling technology under Rockhampton city and Gladstone city climates. Results showed that in both sites, the desiccant cooling system achieved a maximum 1.2 of *COP* when installing 50 m^2 of solar collector's area in the months of November, December, January, February and March. The second best result for the desiccant system under both sites climates was achieved when installing 20 m^2 of solar collectors, recorded an average of 0.99 *COP* for the months of November, December, January, February and March. In addition the absorption cooling system under Rockhampton city and Gladstone city climates achieved a maximum 0.81 of *COP* in the month of January and a minimum of 0.43 of *COP* in the month of July when installing 50 m^2 of solar collector's area. Furthermore the average annual *COP* for the absorption system in both sites recorded near

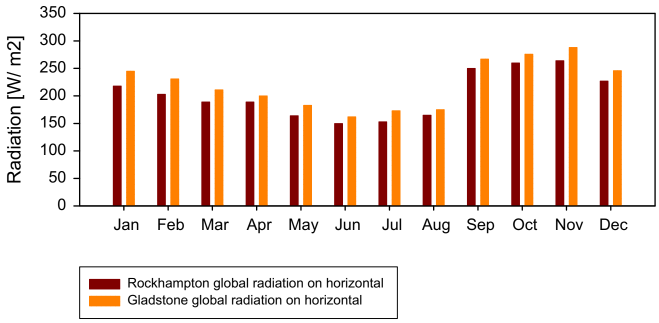

Fig. 6. Rockhampton and Gladstone global radiation on horizontal.

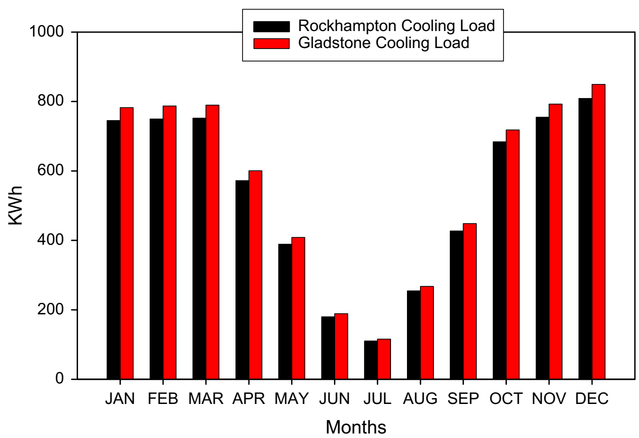

Fig. 7. Building cooling load in Rockhampton and Gladstone.

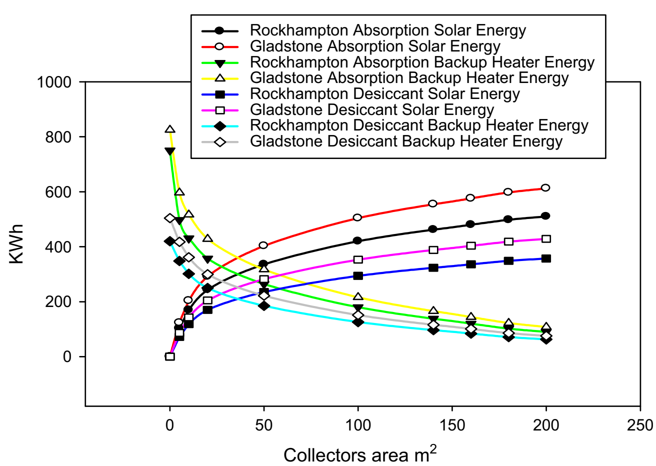

Fig. 8. Absorption and desiccant systems solar energy vs backup heater energy.

0.51, 0.53, 0.58 and 0.62 when installing 5 m^2, 10 m^2, 20 m^2 and 50 m^2 respectively.

Every solar cooling technology requires different amount of primary energy which can be defined as the total electric energy used by the solar cooling system. Figs. 11 and 12 show the primary energy required by the two solar cooling systems (Absorption and Desiccant) under Rockhampton and Gladstone climates considering different solar collectors area. As already noted the primary energy is decreased when increasing the installed solar collector's area. Results showed that the maximum primary energy required was 515 kWh when installing 5 m^2 for an absorption system under Rockhampton climate and 469 kWh under Gladstone climate. For desiccant cooling system the maximum primary energy

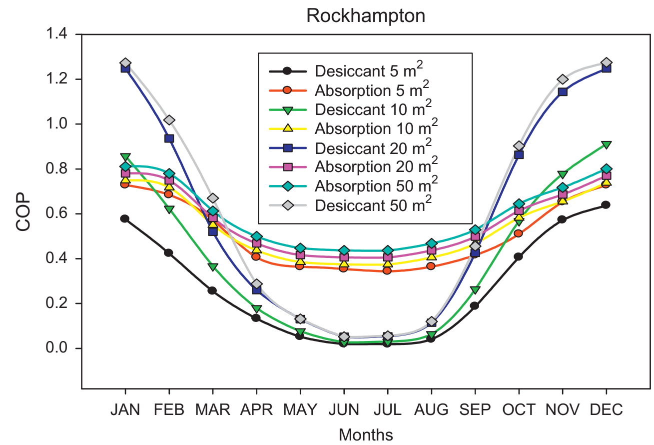

Fig. 9. Rockhampton desiccant and absorption cooling system *COP*.

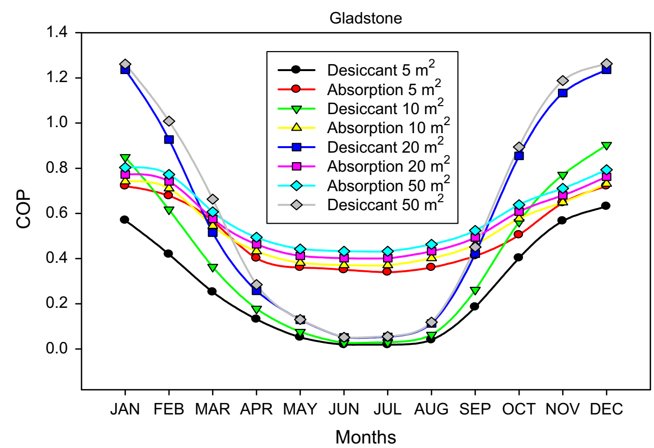

Fig. 10. Gladstone desiccant and absorption cooling system *COP*.

required when installing 5 m^2 of solar collectors area was 491 kWh and 446 kWh under Rockhampton and Gladstone climate respectively. The minimum required primary energy recorded was when installing a desiccant cooling system and 50 m^2 of solar collector's area for at 137 kWh and 150 kWh for Rockhampton and Gladstone sites respectively.

As mentioned earlier the most important technical indicator to assess the viability of any solar cooling system is solar fraction (*SF*). Figs. 13 and 14 show the two sites monthly solar fraction performance using different type of solar cooling systems taken into account installing different solar collector's area. It is found that, the solar fraction was the highest for Rockhampton site in the month of January and December which have the highest cooling load recording near 0.81 of (*SF*) for desiccant cooling system and 0.75 of (*SF*) for absorption cooling system when installing 50 m^2 of solar collector's area.

In the other hand Gladstone site achieved a result of 0.89 *SF* for desiccant cooling system and 0.81 (*SF*) for absorption cooling system in the month of December and the month of January when installing 50 m^2 of solar collector's area. As already noted the solar fraction in both sites was commonly low in May, June, July and August which is the low cooling season. In short the solar desiccant cooling system has a higher solar fraction (*SF*) than solar absorption cooling system under both sites climate.

Fig. 15 shows the variation of solar energy savings using absorption cooling system taken into account different collector's area for

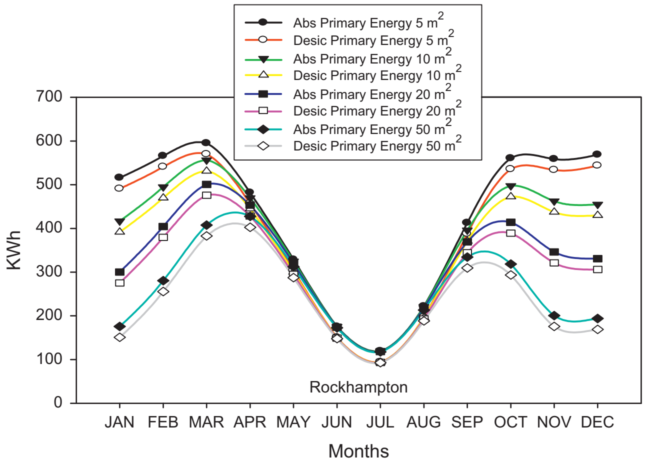

Fig. 11. Rockhampton desiccant and absorption primary energy used.

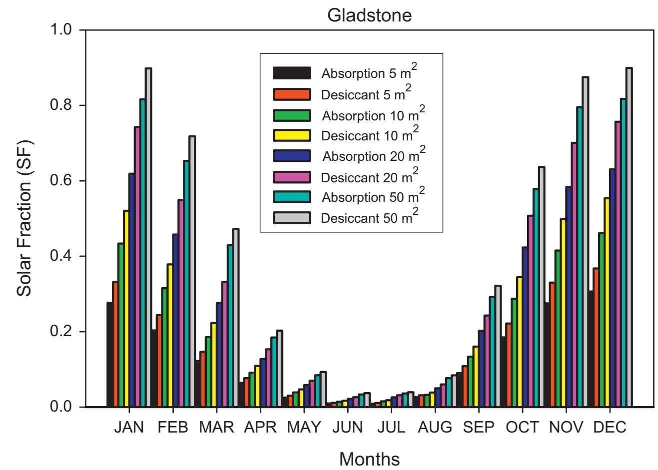

Fig. 14. Gladstone desiccant and absorption systems solar fraction.

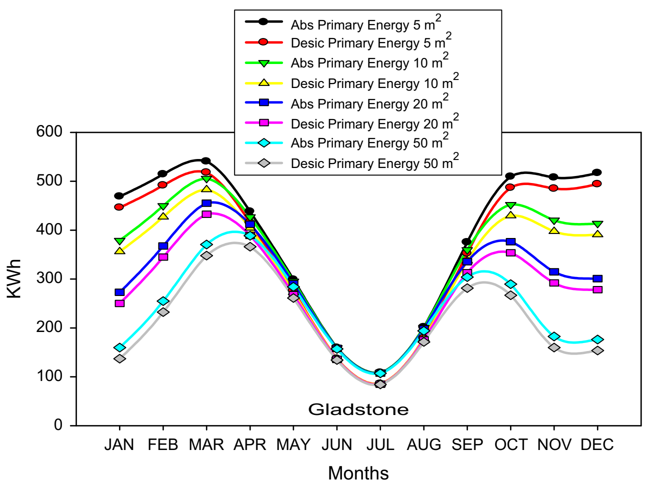

Fig. 12. Gladstone desiccant and absorption primary energy used.

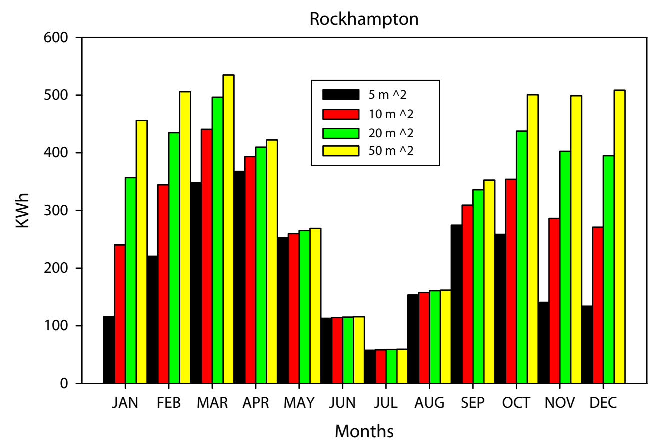

Fig. 15. Energy savings by Rockhampton absorption cooling system.

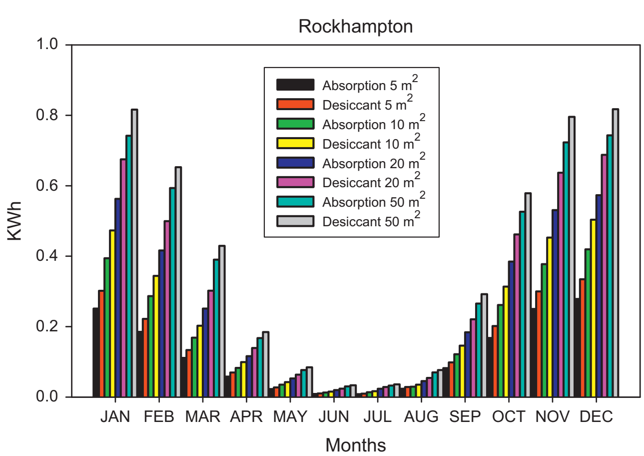

Fig. 13. Rockhampton desiccant and absorption systems solar fraction.

Rockhampton site. Results showed that, installing 50 m^2 of solar collectors will achieve 4385 kWh of energy savings. The maximum saving was in the month of March at 534 KWh then followed by the month of December at 508 kWh and then the month of November at 505 kWh. Moreover the system achieve an annual total of 3867 kWh, 3228 kWh and 2435 kWh when installing 20 m^2, 10 m^2 and 5 m^2 respectively. In addition Fig. 16 shows the variation of energy savings using desiccant cooling system taking into account different solar collector's area. As already noted installing 50 m^2 of solar collectors will achieve 2411 kWh of annual energy savings. The desiccant cooling system in Rockhampton achieved a 2128 kWh, 1775 kWh and 1339 kWh when installing 20 m^2, 10 m^2 and 5 m^2 respectively.

In Gladstone site it was a similar story when compared to Rockhampton site in term of energy savings. Fig. 17 shows the variation of solar energy savings using absorption cooling system taken into account different collector's area for Gladstone site. Results showed that, installing 50 m^2 of solar collectors will achieve 4823 kWh of energy savings. The maximum saving was in the month of March at 589 kWh then followed by the month of December at 559 kWh then the month of February at 556 kWh followed by the month of October, November and January recording 550 kWh, 548 kWh and 501 kWh respectively. Moreover the system achieved an annual total of 3867 kWh, 3228 kWh and 2435 kWh when installing 20 m^2, 10 m^2 and 5 m^2 respectively. In addition Fig. 18 shows the variation of energy savings using desiccant cooling system take into account different solar collector's area for Gladstone site. As already noted installing 50 m^2 of solar collectors will achieve 2652 KWh of annual energy savings. The desiccant cooling system in Gladstone achieved a 2340 kWh, 1954 kWh and 1474 kWh when installing 20 m^2, 10 m^2 and 5 m^2 respectively.

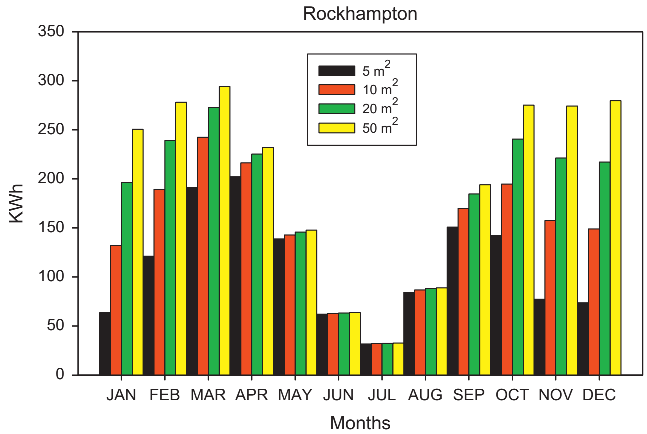

Fig. 16. Energy savings by Rockhampton desiccant cooling system.

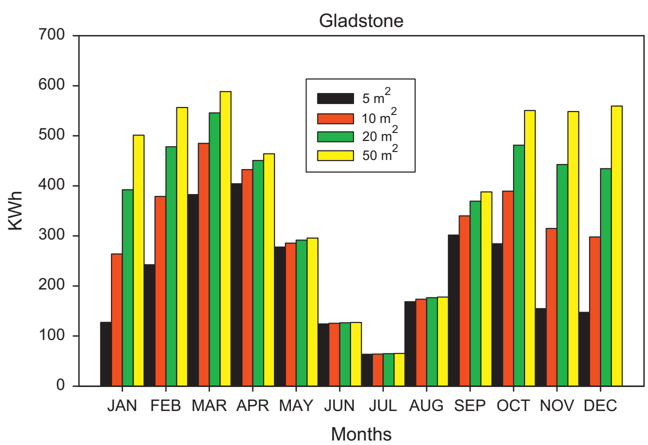

Fig. 17. Energy savings by Gladstone absorption cooling system.

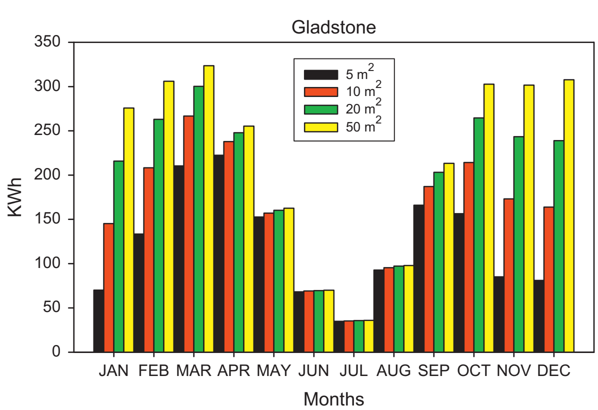

Fig. 18. Energy savings by Gladstone desiccant cooling system.

8. Results validation

To validate results, a comparison analysis has been carried out between simulated results and manual calculation of 5% of simulation time. Manual calculation data was used as input parameters for the TRNSYS simulation model in order to evaluate the performance of the system using the system's performance parameters: solar fraction, coefficient of performance, primary energy and savings energy. The average relative error between simulation and manual calculation data is calculated according to [25,26].

It is clear that there are variations between simulated results and manual calculated values, and that is due to some external causes and other resources such as discrepancies between actual and simulation input in weather data, building operational data and physical properties which were beyond the control of the author. The difference between outdoor actual ambient temperature and simulated temperature varies between ±2% and ±4%, while the difference between actual and simulated outdoor relative humidity is ±3%. In addition, the difference between simulated and manual calculated system parameters namely: system *COP*, primary energy used, energy savings and solar fraction was ±6%, ±9%, ±7% and ±9% respectively.

9. Conclusion

In this paper a detailed comparative study between two different solar cooling technologies; absorption cooling and desiccant cooling for an institutional building in Central Queensland, Australia is presented. The study identified each technology performance indicator for assessing the viability of each system under two Central Queensland cities (Rockhampton and Gladstone) taking into account installing different solar collector's area. The procedure developed in this study can be applied by designers to evaluate the viability of using solar cooling technology. Results showed that the proposed building under Gladstone climate has 6750 kWh of total annual cooling load. The same proposed building under Rockhampton climate has a 6428 kWh of total annual cooling load. In both sites, using a desiccant cooling system achieved a near 1.2 coefficient of performance (*COP*) during the high cooling demand season when installing 50 m^2 of solar collector's area and an average of 0.99 of *COP* when installing 20 m^2 of solar collector's area during the cooling season. While using absorption cooling system in both sites achieved an average near 0.75 of *COP* during the cooling season when installing 50 m^2 and near 0.6 of *COP* during the cooling season when installing 20 m^2 of solar collector's area. Due the existence of backup heater and other electrical components like water pumps and blowers within solar cooling systems, primary energy is required. Results showed that using absorption cooling system in both sites will draw more primary energy than when using desiccant cooling system. The total annual primary energy drawn by the absorption system was 5104 kWh, 4588 kWh, 3948 kWh and 3155 kWh when installing 5 m^2, 10 m^2, 20 m and 50 m^2 respectively under Rockhampton climate. While Gladstone absorption cooling system primary energy required was 4640 kWh, 4170 kWh, 3590 kWh and 2868 kWh when installing 5 m^2, 10 m^2, 20 m^2 and 50 m^2 respectively. Desiccant cooling system under both sites achieved a higher solar fraction SF than absorption cooling system. The maximum was recorded in the month of December at 0.89 and 0.82 for Gladstone and Rockhampton respectively. Thus using absorption cooling technology in Gladstone sites achieved the highest annual energy savings at 4822 kWh, 4225 kWh, 3551 kWh and 2679 kWh when installing 50 m^2, 20 m^2, 10 m^2 and 5 m^2 respectively. According to the Australian government, Office of climate change, the average annual temperature in Central Queensland region has increased 0.5 °C in the last 10 years and will reach up to 4.5 °C in the year 2070. Replacing conventional HVAC system with the ones can be powered by renewable energy can be a solution for the climate change dilemma despite the fact of high installing cost.

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